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### NORTHERN CLIMATE HEAT PUMP LABORATORY EVALUATION

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One of the seven prototype CEA Northern Climate Heat Pumps was installed and tested in Ontario Hydro's Heat Pump Laboratory. Tests show that the steady-state coefficient of performance figures are 15% to 20% higher than those of a commercially-available model tested in 1985. Low-temperature heating performance surpassed that of all current air-source heat pumps on the market.

#### 1. INTRODUCTION

Residential heat pumps not designed for service in a cold climate often suffer from performance and reliability problems when they are installed and operated under Canadian wintertime conditions. To overcome these problems, the Canadian Electrical Association (CEA) and Ontario Hydro jointly sponsored a program to design a heat pump suitable for northern climates. A 1980 Ontario Hydro Research Division (OHRD) report written for the CEA on the "Optimization and Evaluation of a Northern Climate Residential Air-Source Heat Pump" describes many improvements over the (then current) models/1/.

Based on the encouraging results of the above program, CEA provided funds to build commercially-viable models incorporating the design changes recommended in the OHRD report. A Canadian manufacturer was selected and seven prototypes of the CEA heat pump were built. One of these seven new high-performance heat pumps was installed in the Ontario Hydro environmental chamber for laboratory evaluation.

The laboratory evaluation was carried out with three objectives in mind:

- 1) evaluate the performance of the unit,
- 2) conduct defrost tests, and
- 3) suggest possible improvements to the heat pump.

The laboratory evaluation was conducted in the latter half of 1985, and additional information became available from field test sites of the other six units during the '85 - '86 and '86 - '87 winter seasons.

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## 2. HEAT PUMP DESCRIPTION

### 2.1 General

The prototype heat pump is a residential, split-system, air-source design with a nominal heating capacity of 9 kW (2.5 ton). A microprocessor controls the operation of the heat pump and three status lights indicate the condition of the heat pump system. Further details and a schematic of the heat pump can be found in Appendix A.

### 2.2 Improvements Embodied in the Heat Pump

This section will list a few of the improvements embodied in the CEA heat pump. These design changes were based on recommendations from the OHRD report.

1. A thermal-electric expansion valve and a microprocessor-based control work together to provide optimal refrigerant control over a wide range of outdoor temperatures. This system maintains the outdoor evaporator under fully-flooded (most efficient) conditions at all times during the heating mode.
2. An indoor compressor location eliminates certain heat losses and improves the service and maintenance environment. The indoor compressor and control location eliminates the need for a crankcase heater for cold weather start-up.
3. The heat pump has large heat exchangers. This results in greater heat transfer efficiency and capacity.
4. A defrost bypass valve improves heat and mass flow to the outdoor coil to reduce defrost cycle time.
5. An accumulator heat exchanger improves the transient response of the system during start-up and defrost recovery.
6. The heat pump uses demand defrost to minimize unnecessary defrost cycles.

These improvements are discussed in greater detail in Appendix A.

## 3. TESTING AND RESULTS

Steady-state performance testing at a variety of simulated outdoor conditions was conducted in the OHRD heat pump test facilities. Descriptions of the tests carried out and the observations made on the heat pump as delivered by the manufacturer are contained in Appendices B, C, D and E.

The most significant results were as follows:

1. The CEA prototype heat pump yields superior heating performance, especially at low temperatures, making it ideal for cold climate applications (ie, Ontario).
2. Compared to a typical (1985) heat pump, the CEA prototype performs 14% better at an outdoor temperature of +8.3°C and 20% better at -8.3°C (Figure 1), and an impressive 40% better at -25°C.
3. The performance improvement will mean reduced demand at winter peak times. From the performance shown in Figure 2, it can be seen that if the heat pump was installed in an area with a design temperature of -20°C, it would contribute 1.7 kw of "free" heat at this temperature. If such a machine were installed in a house with a -20°C heat loss of 8 kw (as would be typical in today's market for a 2.5 ton heat pump) the 1.7 kw demand saving would amount to 21%. If a larger machine were installed in the same house (as might be done under different pricing or strategic conservation scenarios), the demand savings could range upwards to 46%.
4. Even at temperatures as low as -25°C, the CEA heat pump continues to produce a substantial heat gain (unlike most commercial heat pumps) as shown in Figures 1 and 2.
5. It was not possible to accurately assess the cooling performance since the test heat pump appeared to have a faulty component in the cooling circuit.
6. The pressure-initiated defrost system installed in the prototype heat pump could experience problems during field application. The original OHRD work recommended against the use of this type of defrost control. There are more reliable types of defrost systems available.

4. CONCLUSIONS

The prototype heat pump has the best low-temperature heating performance in the world and, consequently, it will require less electric resistance back-up heat in cold weather. The problems encountered with defrost and cooling are minor and easily solved.

The availability of such a heat pump on the market will provide benefits for the homeowner, for Ontario Hydro and for the province of Ontario.

- The homeowner gains by reducing the cost of heating his house.
- Ontario Hydro gains by having a high performance product to improve our competitive position and credibility with the public. On the supply side, we gain by having a product with an improved load characteristic. On the demand side, there is potential for energy conservation. Demand savings up to 46% could be achieved for a standard electrically heated home.

- Ontario gains by producing a world-leading product which could be exported to other cold climate areas like the northern USA, northern Europe, and northern Japan.

## 5. RECOMMENDATION

The technology for improved heat pumps of the type described in this report is now available for Ontario Hydro to market.

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2. "Method of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment", Standard No. 37-1978 (The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., New York, N.Y.).
3. "Standard for Unitary Heat Pump Equipment", Standard No. 240-76, (The Air Conditioning and Refrigeration Institute, Arlington, VA.).
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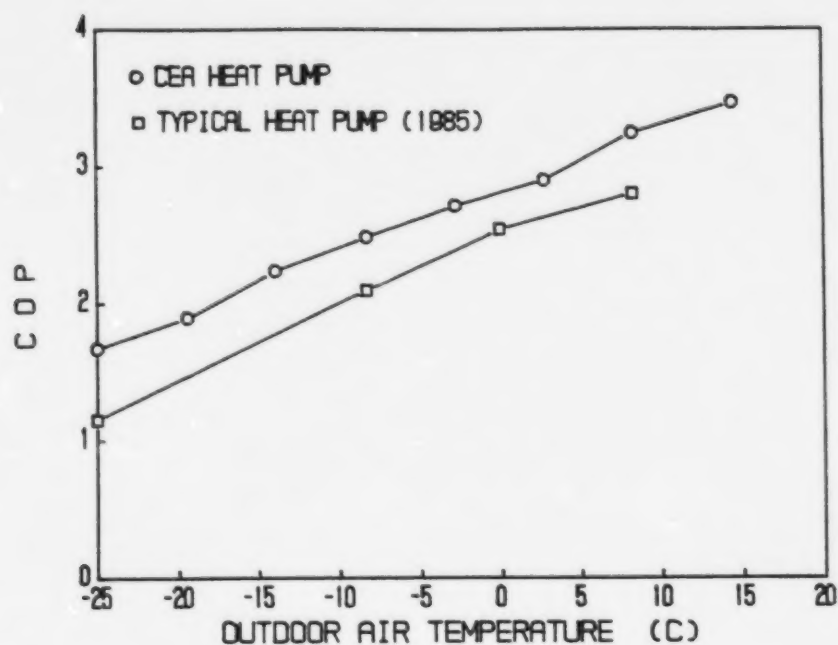


FIGURE 1. HEAT PUMP PERFORMANCE

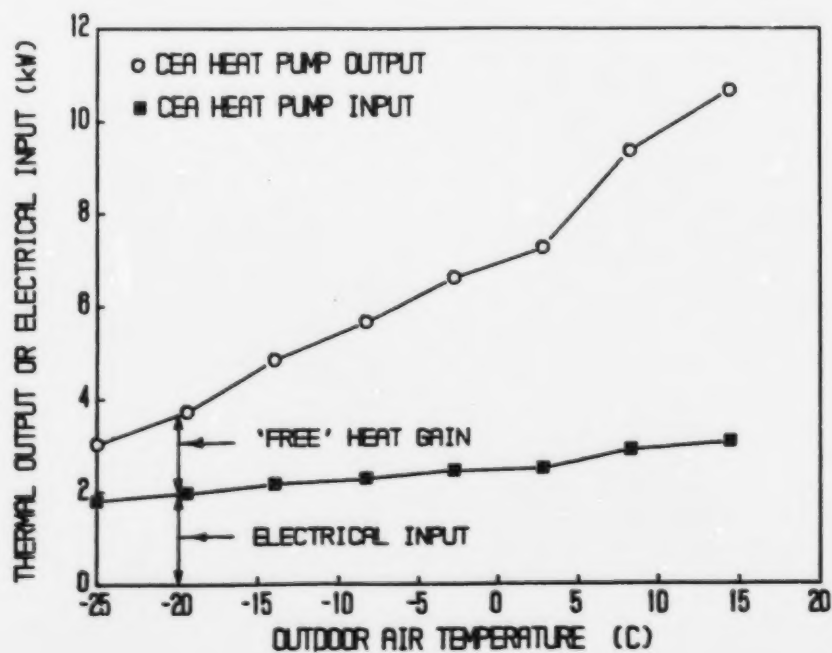


FIGURE 2. HEAT PUMP THERMAL OUTPUT AND ELECTRICAL INPUT

## APPENDIX A

### HEAT PUMP DESCRIPTION

#### General

The prototype heat pump is a residential split-system, air-source design with a nominal heating capacity of 9 kW (2.5 ton). For added efficiency and easy service access, the compressor is located indoors. The airflow through the large indoor coil is handled with a variable-speed 1/4 hp direct-drive blower. Refrigerant control is carried out by a microprocessor-controlled thermal-electric expansion valve which receives input from the discharge line thermostat, room thermostat, defrost control and receiver level thermistor, and controls the operation of the compressor, reversing valve, bypass valve, and expansion valve. The microprocessor board has three system status lights to indicate the condition of the heat pump system. A failure of the microprocessor board results in the loss of proper heat pump operation. However, a level of comfort can still be maintained as the microprocessor is not necessary to operate the back-up electric heaters. A heat pump schematic is presented in Figure 3.

The heat pump is factory-charged with R-22 refrigerant. The indoor and outdoor units contain 3.6 kg (8 lbs) and 0.9 kg (2 lbs) of charge respectively. Precharged refrigerant tubing is also supplied with the heat pump. The fittings are "one shot" type and care is required before the final tightening and puncturing of the diaphragm.

#### Improvements Embodied in the Heat Pump

This section will list a few of the improvements embodied in the heat pump. These design changes were based on recommendations from the OHRD report/1/.

- 1) The compressor is located indoors thus avoiding the need for crankcase heating and allowing shell and discharge piping losses to contribute to useful heat pump output. The indoor location also improves the service and maintenance environment.
- 2) The unit uses demand defrost. However, the method of defrost initiation differs from the design described in the OHRD report. The initial design called for sensing an increase in outdoor fan current but the present prototype utilizes a defrost control unit that is preset to defrost the outdoor coil whenever these conditions occur simultaneously:
  - outdoor coil temperature below  $-2.8^{\circ}\text{C} \pm 1.7^{\circ}\text{C}$  ( $27^{\circ}\text{F} \pm 3^{\circ}\text{F}$ )
  - outdoor coil static pressure drop greater than  $50 \pm 1.5$  Pa ( $0.20" \pm 0.03"$  WC)
  - after an initiation inhibit period of  $20 \pm 2$  minutes has elapsed since the previous defrost terminated.

Defrost is terminated either when the coil temperature reaches  $12.7^{\circ}\text{C} \pm 2.1^{\circ}\text{C}$  ( $55^{\circ}\text{F} \pm 5^{\circ}\text{F}$ ), or after a  $10 \pm 1$  minute duration whichever occurs first.

- 3) A thermal electric expansion valve, an accumulator heat exchanger, and a special receiver work together to provide optimal refrigerant control. Receiver level control ensures that the outdoor evaporator remains flooded and a fixed portion of the condenser is devoted to subcooling over the entire range of operating conditions. A thermistor senses the receiver level and through the help of the microprocessor delivers a signal to the expansion valve. Should the receiver level be high, the expansion device responds by opening, thus permitting the receiver level to fall. In addition to providing optimal flow conditions, the voltage setting for the expansion valve position can be stored in memory for fast system recovery after a defrost or off cycle.
- 4) The heat pump utilizes an accumulator heat exchanger which performs the following useful functions.
  - it enhances transient performance during start-ups and following defrost cycles
  - it allows for oscillations in the expansion valve opening operations without starving the evaporator
  - it helps to ensure that all of the evaporator circuits remain flooded.
- 5) Coil areas of both the indoor and outdoor unit are large and at the higher end of normal practice. This results in greater heat transferred by the system.
- 6) A defrost bypass valve, located across the cooling mode expansion device (capillary tubes), helps prevent oil foaming in the compressor sump and maintain a higher rate of mass flow and heat delivery to the outdoor coil during defrost cycles. This reduces the defrost cycle time compared to defrosts where only the cooling mode expansion device is used.

The above improvements all work to provide a higher performance heat pump.

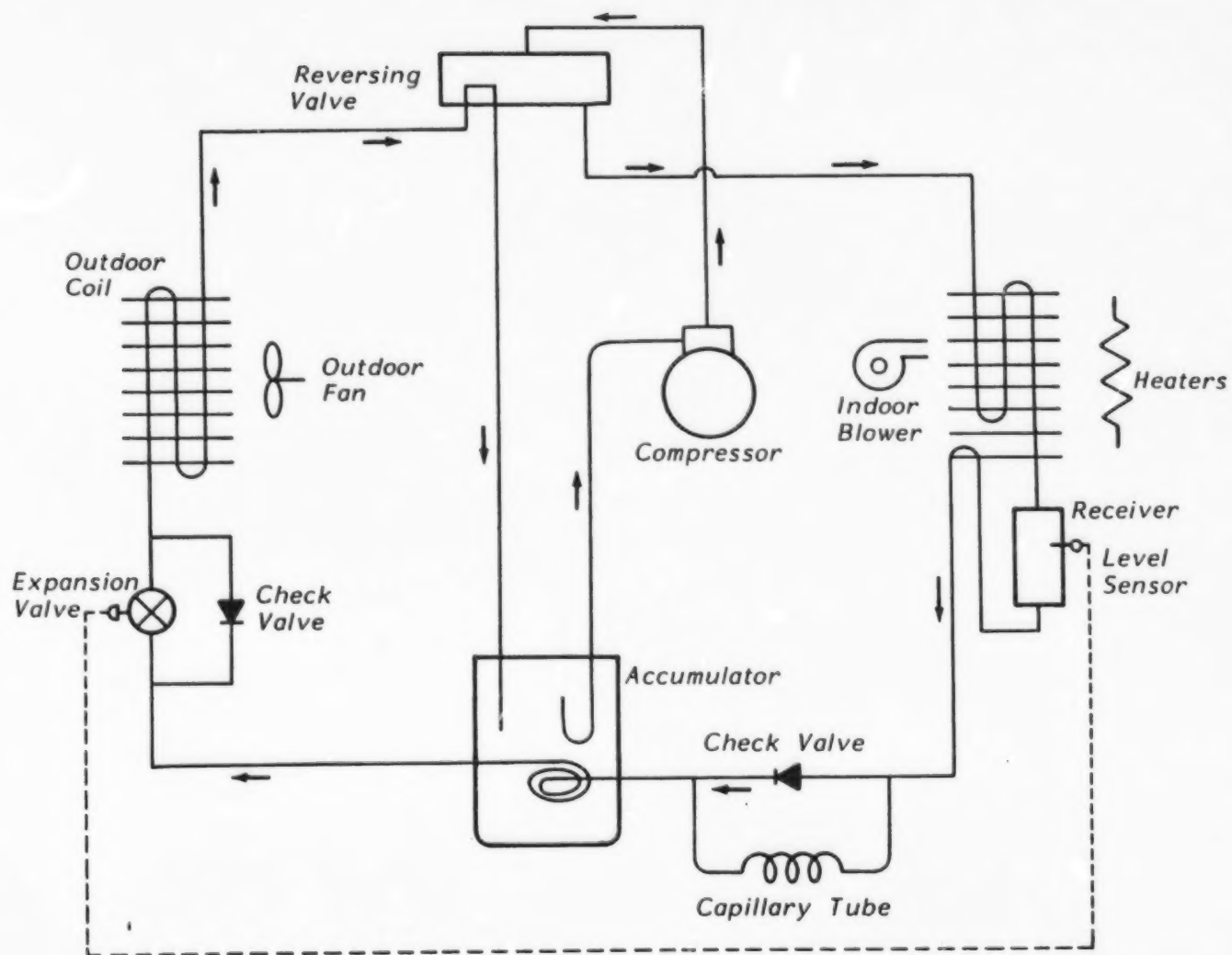


FIGURE 3  
HEAT PUMP BLOCK DIAGRAM  
(HEATING MODE)

## APPENDIX B

### LABORATORY SET-UP

The heat pump was evaluated using the "indoor air enthalpy method" described in ASHRAE standard 37-78/2/. The equipment tested was located in two rooms (see Figure 4). Suitable means for maintaining the rooms at desired temperatures are provided. On the indoor side, the heat pump discharges into a reconditioning loop which maintains the test room at desired conditions. The outdoor unit discharges directly into the outdoor room.

Air flow rate was measured by the nozzle apparatus consisting of a receiving chamber, discharge chamber and one or more nozzles (see Figure 4). Capacities were determined from measurements of entering and leaving wet and dry bulb temperatures and the associated air flow rate.

High and low temperature tests were conducted at  $\pm 8.3^{\circ}\text{C}$  (47F and 17F) as prescribed in ARI-240/3/. The outdoor room was kept at 80% relative humidity for all heating tests. COP readings were taken every 10 minutes until seven consecutive readings were attained.

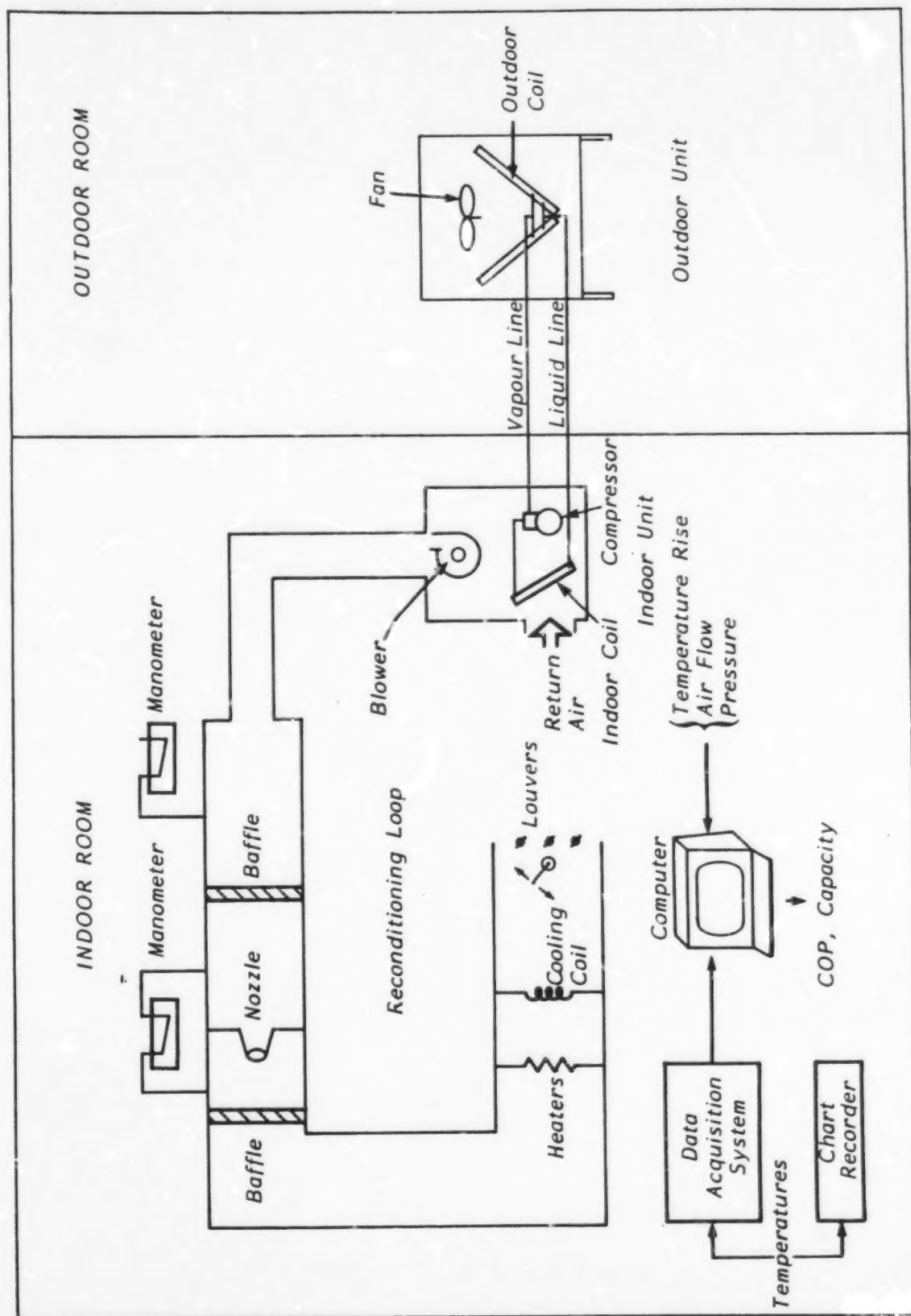


FIGURE 4  
EXPERIMENTAL SET-UP

## APPENDIX C

### ASSEMBLY DEFECTS IN PROTOTYPE UNITS

During the laboratory set-up and operational start-ups, certain assembly defects were noted in the hand-built prototype.

#### Airflow

The indoor blower of the heat pump in the laboratory was wired to run backwards. A check on the other units revealed two more heat pumps with the same problem. This was corrected by a simple wiring change.

A similar problem also existed with the outdoor fan. The dual speed outdoor fan is designed to operate at high speed at ambients lower than  $13.9^{\circ}\text{C} \pm 3.3^{\circ}\text{C}$  and at low speed at ambients higher than  $22.2^{\circ}\text{C} \pm 3.3^{\circ}\text{C}$ . On at least 4 of the 6 units, the fan speed setting was reversed. The fan wires had to be reconnected according to the wiring schematic.

#### Refrigerant Charge

For any heat pump system, correct charge is a key for proper heat pump operation. At least two of the six prototypes had indoor units undercharged by 2.5 kg (5.5 lb). This was traced to incorrect charging labels pasted on the indoor unit cabinet.

#### Defrost Sensor

On some of the defrost sensor units, the defrost control wires were incorrectly placed. Thus, a contact closure would not be acknowledged by the microprocessor. In addition, the plastic hoses for determining the air side pressure drop across the coil were also reversed. This meant that a defrost cycle could never be initiated. Simple adjustments rectified the problem.

#### Microprocessor Board

A list of problems were spotted with the microprocessor board.

- i) There were large amounts of flux left near the solder joints which should be cleaned to prevent high humidity causing current leakage problems.
- ii) A large number of jumpers on the component side of the microprocessor board were unacceptably close to other metal conductors. Insulation should be added.
- iii) The board requires a conformal coating for protection against debris and moisture.

iv) Some of the crimp connectors above the board were too large for the wire sizes used. Also, several bad crimp connectors needed replacement.

v) Some of the plastic snap fasteners for the board were broken.

These assembly defects were considered minor and were easily corrected. However, these defects reflect poor workmanship and would be a source of customer dissatisfaction if carried over to a production unit.

## APPENDIX D

### OPERATING DEFECTS IN PROTOTYPE UNITS

The following defects appear with the heat pump in operation.

#### Microprocessor Board

- i) There is no transient protection on any of the inputs, outputs or at the power supply. This is unacceptable as normal transients present in a house could damage components. Laboratory tests with a standard transient generator destroyed the output triacs on the board.
- ii) In the heating mode, the microprocessor has two modes of operation: "initial" and "normal". The initial program mode is set after an off cycle or a defrost cycle. The normal program mode runs only after the receiver level goes through one complete up and down cycle. However, this occurred for only a range of outdoor temperatures (between +8°C and -14°C). Outside this temperature range the heat pump had great difficulty in coming out of the initial start-up mode because of restricted range of valve openings imposed by the microprocessor algorithm during start-up. This results in poor refrigerant control and lower output and efficiency. The initial program mode needs to be refined to handle start-ups and defrost recovery for a wider temperature range.

For the laboratory evaluation, it was necessary to "fool" the microprocessor into the normal program mode by controlling the voltage to the expansion valve. At high outdoor temperatures, the receiver level rises but never comes back down. To force the receiver level down, apply increasing voltage (up to 24 volts) to the expansion valve until the receiver level drops and then remove this external power supply. A drop in the receiver level can be measured with a DVM connected across terminals RT and XT. The DVM will measure around 10 V for a high receiver and about 3 V for a low receiver. The end of the initial program mode will be indicated by the stoppage of the flashing status lights.

To force the heat pump into a normal heating mode at low outdoor temperatures disconnect the wire to the expansion valve until the receiver level rises. Re-connect the expansion valve and when the heat pump goes into the normal heating mode, the status lights will stop flashing.

#### Control Circuit Overload

The operational checks carried out soon after the installation of the heat pump revealed another design flaw. A 3.2 A fuse for the heat pump controls burned out more than once during start-ups. A check of the control wiring revealed a high operating load at that terminal location. This problem was solved by dividing the load and installing an additional fuse.

### Defrost Sensor

The defrost sensor pressure initiation setting was incorrectly set for proper operation. A check on the defrost sensor revealed the pressure indicator setting to be too low. A coil static pressure drop of 0.17" wg, corresponding to only 29% coil coverage, triggered the defrost sensor. Moreover, we regard the set coil static pressure drop of 0.2" wg as being too low. The percentage coverage of the coil would be only 38% for defrost initiation. A more practical pressure initiation setting would correspond to 50% to 75% coil coverage. The defrost sensor in the laboratory was later calibrated to trigger at 0.24" wg static pressure or 60% coil coverage. This represented a reasonable compromise between the two limits.

### Summary

These operating defects are relatively easy to fix. The major change required is the refinement to the microprogram algorithm to handle start-ups and defrost recovery outside the +8° to -14°C temperature band.

An additional defrost sensor problem was noted. The particular version of the commercially-available sensor installed in the prototype has a built-in 6-hour defrost initiation feature. This feature would normally be employed to activate a defrost every 6 hours of compressor run time if no other event had triggered a defrost. However, as wired in the prototype, the defrost control has no perception of the compressor being off, and thus it issues a defrost command every six hours of clock-elapsed time. If the heat pump was actually off when this signal was received, the microprocessor interpreted this state as "abnormal" and locked out the heat pump until manually reset. This defect was corrected by the manufacturer through a revision to the microprocessor program.

## APPENDIX E

### HEAT PUMP PERFORMANCE

#### Heating Mode

##### Standard Test Conditions

These tests were carried out using a method described in ASHRAE standard 37-78/2/. High and low temperature heating tests were conducted at  $\pm 8.3^{\circ}\text{C}$  ( $47^{\circ}\text{F}$  and  $17^{\circ}\text{F}$ ) as prescribed in ARI-240/3/. Tests were also conducted for several other outdoor temperatures. For all tests, the return inlet air was kept constant at  $21.1^{\circ}\text{C}$  ( $70^{\circ}\text{F}$ ) and the outdoor air was maintained at 80% relative humidity.

High and low temperature heating tests yielded coefficient of performance (COP) values of 3.2 and 2.5 respectively. Performance test results for several outdoor temperatures are shown in Figure 5. These tests were carried out with the microprocessor in the "normal" program mode.

The laboratory COP values agree with the manufacturer's published data. Figure 5 also shows the performance results of a commercially available air source heat pump (1985 model)/4/. At an outdoor temperature of  $+8.3^{\circ}\text{C}$  ( $47^{\circ}\text{F}$ ), the prototype northern climate heat pump has a COP value which is 14% higher than the 1985 heat pump model. The improvement is even greater at lower temperatures. At  $-8.3^{\circ}\text{C}$  ( $17^{\circ}\text{F}$ ) the prototype unit has a COP value 20% more than the popular 1985 heat pump model/4,5/. At  $-25^{\circ}\text{C}$  ( $-13^{\circ}\text{F}$ ), the prototype unit has a COP of 1.7. This means that at  $-25^{\circ}\text{C}$ , the CEA heat pump continues to produce a heat gain (unlike most commercial heat pumps).

The prototype heat pump yields superior heating performance results. The unit performs especially well at low temperatures, thus making it ideal for northern climate application.

##### Test With Different Indoor Air Temperatures

Tests were carried out with constant outdoor conditions but different indoor air temperatures. The outdoor air was maintained at  $-2.8^{\circ}\text{C}$  ( $27^{\circ}\text{F}$ ) and 80% relative humidity. Figure 6 shows the increase in COP with lower indoor air temperatures.

##### Test With Different Outdoor Fan Speed

The outdoor fan has two speed settings, high and low. At low speed, the measured static pressure drop across the frost-free coil is 0.05" of water. At high speed, the static pressure is 0.1" of water.

Performance tests were carried out under similar test conditions for the two fan speeds. The outdoor air was maintained at  $-2.8^{\circ}\text{C}$  ( $27^{\circ}\text{F}$ ) and 80% relative humidity. Only a very slight change (0.7%) was noticed between the two COP

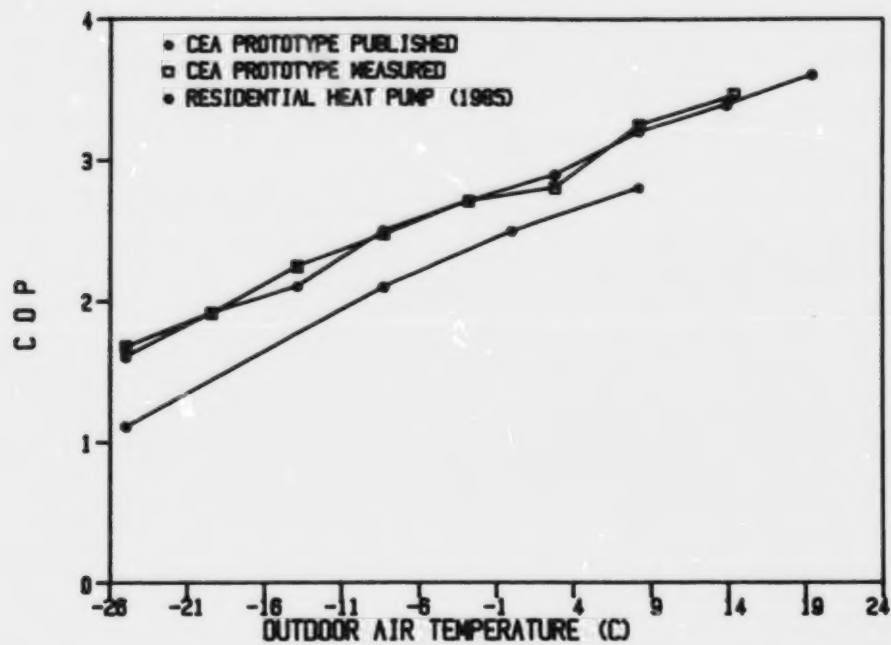


FIGURE 5. HEAT PUMP PERFORMANCE IN THE HEATING MODE

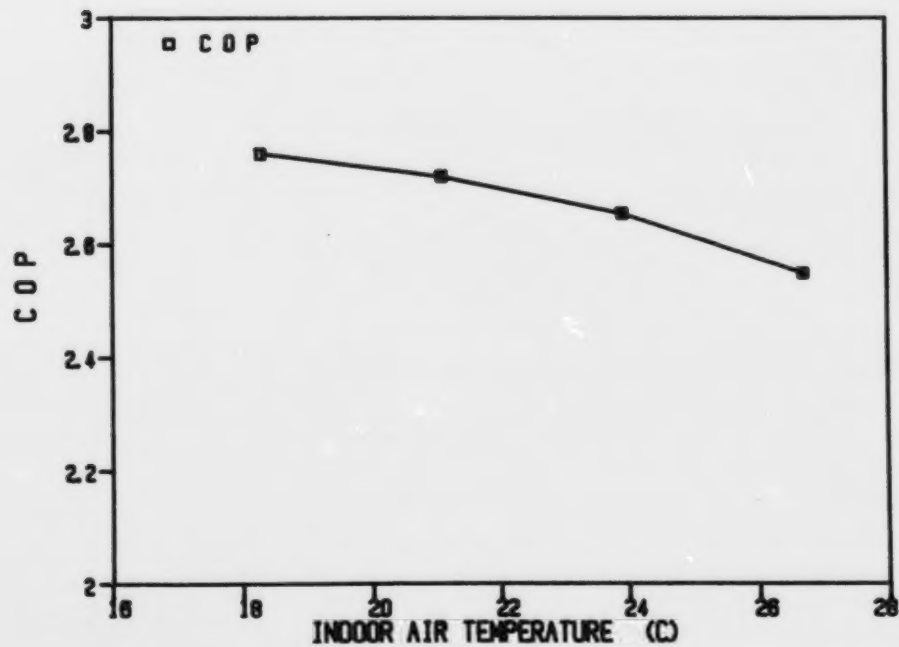


FIGURE 6. EFFECT OF INDOOR AIR TEMPERATURE ON HEAT PUMP PERFORMANCE

values. However, the capacity of the heat pump with the faster fan speed was higher by 277.5 watts. Fan power itself was 260 W (high) and 190 W (low).

Table 1. Heating Test With Two Outdoor Fan Speeds

| Fan Speed | Static Pressure<br>("wg) | COP  | Capacity<br>(W) |
|-----------|--------------------------|------|-----------------|
| high      | 0.1                      | 2.71 | 7036.5          |
| low       | 0.05                     | 2.73 | 6759.0          |

#### Cooling Mode

The prototype heat pump is reversible and tests were also carried out in the cooling mode. Cooling performance is customarily calculated by dividing the cooling capacity in Btu/h by the power input in watts for any given set of conditions. This Energy Efficiency Ratio (EER) is expressed in Btu/h per watt.

Commercially-available air-conditioners and heat pumps normally have EER ratings between 7 and 11/5/. Higher EER ratings indicate better cooling performance. The prototype heat pump yielded poor cooling performance results. The EERs are listed in Table 3 for different outdoor and indoor temperatures.

Table 2. Cooling Mode Results

| Outdoor<br>Temperature<br>(°C) | Return Inlet<br>Air Temperature<br>(°C) | Cooling<br>Capacity<br>(W) | EER<br>(Btu/h)<br>(W) |
|--------------------------------|-----------------------------------------|----------------------------|-----------------------|
| 35                             | 21.1                                    | 4318.5                     | 4.59                  |
| 32.2                           | 23.9                                    | 5527.8                     | 6.16                  |
| *35                            | *26.7                                   | 5847.4                     | 6.24                  |

\* These conditions match ARI conditions most closely /5/.

The low EER values may be due to a leaky check valve. The check valve in question is located across the capillary tubes in the indoor unit. The function of the check valve is to force passage of the refrigerant through the cooling mode expansion device (ie capillary tubes). An inspection revealed possible unwanted refrigerant flow through this check valve during cooling

mode. Consequently, the cooling performance may have suffered because the capillary tubes were not properly controlling refrigerant flow.

#### Defrost Mode

The defrost cycle is triggered by a commercially-available sensor which measured coil static air-pressure drop, coil temperature, and time since the last defrost. Details are in Appendix A. Once initiated, the defrost cycle terminates when coil temperature reaches  $+12.7^{\circ}\text{C} \pm 2.1^{\circ}\text{C}$  or when the 10 minute limit is up.

Defrost cycles initiated in the laboratory usually lasted less than the 10 minute timer limit. Defrost recovery periods were around 9 to 12 minutes at temperatures near freezing (where frosting occurs readily). The prototype heat pump experienced difficulty with defrost cycles initiated outside the microprocessor program operating range (i.e.  $+8^{\circ}\text{C}$  to  $-14^{\circ}\text{C}$ ). A refinement to the microprogram is necessary to handle recovery after defrost for a wider temperature band.

Tests also revealed the pressure initiation setting in the defrost sensor to be too low. The recommended coil static pressure drop of 0.2" of water corresponded to only 38% coil blockage. A more practical pressure initiation setting would correspond to 50% to 75% coil blockage. The defrost sensor in the laboratory was later calibrated to trigger at 0.24" of water static pressure or 60% coil blockage. This represented a reasonable compromise between the two limits.

This particular type of defrost sensor may develop problems during field application. The flexible diaphragm in the pressure switch can harden at lower temperatures leading to erratic operation. Moreover, the air passages for sensing the coil static pressure drop can become blocked with frost or from the collapse of the plastic hoses. Stronger tubing is recommended to prevent blockage of the air passages due to bends. It should be remembered that the original CEA report /1/ recommended against pressure initiated defrost for the reasons noted above.

#### Heating Test With R-502 Refrigerant

Heating tests with R-502 refrigerant were carried out to compare performance characteristics with that of R-22 refrigerant. R-502 is an azeotrope of R-22 and R-115. It has better low-temperature characteristics but care must be taken when utilizing the heavier R-502 with a compressor designed to operate with R-22. The purpose of this test was to note any improvement in heat pump performance or capacity.

Figures 7 and 8 show the COP and the capacity for several outdoor temperatures. The R-502 heat pump performance as indicated by the COP showed little or no change from those plotted in Figure 1 for R-22. However, an approximate 10% increase in capacity was noted when using R-502. The higher capacity is due to the higher heat transfer capability of the heavier R-502 refrigerant, especially at low temperatures.

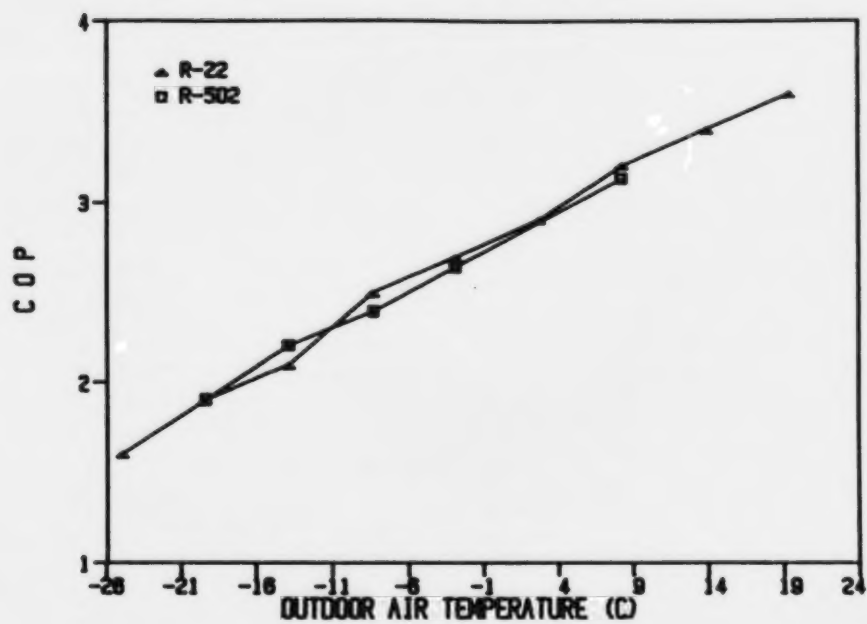


FIGURE 7. PROTOTYPE HEAT PUMP PERFORMANCE

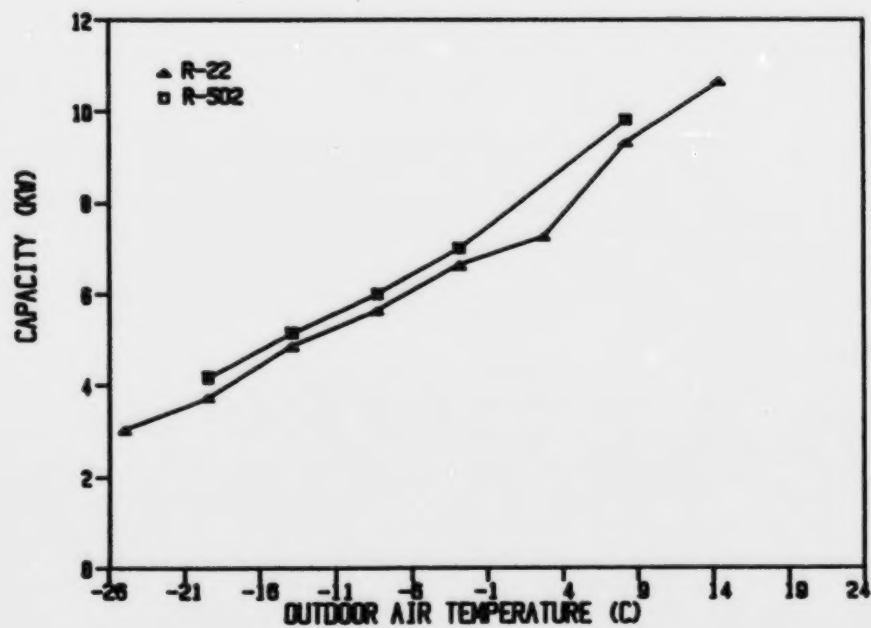


FIGURE 8. PROTOTYPE HEAT PUMP CAPACITY